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DNA-TR-84-394

AD-A162 704

FEASIBILITY STUDY OF A FAST ACTING VALVE FOR THE LB/TS

Sverdrup Technology, Inc.
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1 November 1984

Technical Report

CONTRACT No. DNA 001-84-C-0197

Prepared for
Director
DEFENSE NUCLEAR AGENCY
Washington, DC 20305-1000

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188
Exp. Date: Jun 30, 1986

1a. REPORT SECURITY CLASSIFICATION	1b. RESTRICTIVE MARKINGS
UNCLASSIFIED	

2a. SECURITY CLASSIFICATION AUTHORITY	3. DISTRIBUTION/AVAILABILITY OF REPORT
	Approved for public release; distribution is unlimited.
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE	
N/A since UNCLASSIFIED	

4. PERFORMING ORGANIZATION REPORT NUMBER(S)	5. MONITORING ORGANIZATION REPORT NUMBER(S)
	DNA-TR-84-394

6a. NAME OF PERFORMING ORGANIZATION	6b. OFFICE SYMBOL (If applicable)	7a. NAME OF MONITORING ORGANIZATION
Sverdrup Technology, Inc		Director Defense Nuclear Agency

6c. ADDRESS (City, State, and ZIP Code)	7b. ADDRESS (City, State, and ZIP Code)
P.O. Box 884 Tullahoma, TN 37388	Washington, DC 20305-1000

8a. NAME OF FUNDING/SPONSORING ORGANIZATION	8b. OFFICE SYMBOL (If applicable)	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER
		DNA 001-84-C-0197

8c. ADDRESS (City, State, and ZIP Code)	10. SOURCE OF FUNDING NUMBERS			
	PROGRAM ELEMENT NO.	PROJECT NO.	TASK NO.	WORK UNIT ACCESSION NO.
	62715H	G55AMXG	R	DH008365

11. TITLE (Include Security Classification)

FEASIBILITY STUDY OF A FAST ACTING VALVE FOR THE LB/TS

12. PERSONAL AUTHOR(S)
S.P. Crump and R.F. Starr

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Technical	FROM 840420 TO 841001	1984, November 1	58

16. SUPPLEMENTARY NOTATION
This work was sponsored by the Defense Nuclear Agency under RDT&E RMSS Code B345084466 G55AMXGR00022 H2590D.

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Impulse Facilities Blast Wave Shaping
14	2		Fast Acting Valves Starting Devices
13	11		Sliding Sleeve Valve Diaphragm Substitutes

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

This study explores the feasibility of using a sliding sleeve concept to produce a high-pressure, large-diameter valve which can be rapidly opened or closed. It was studied for application as a starting device at the outlet of a pressurized driver tube in a shock down type wind tunnel. Charge pressures of 2500 psi, valve sizes equivalent to a 36-inch diameter clear opening and opening times of 50 milliseconds or less were found to be obtainable. A valve with these characteristics was conceptually designed. Successful experience with smaller valves based on the same concept and problems associated with scaling up these designs are discussed. A development program which addresses these problems is proposed, and current costs for such a valve and development program are established.

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT	21. ABSTRACT SECURITY CLASSIFICATION
☐ UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT. ☐ DTIC USERS	UNCLASSIFIED

22a. NAME OF RESPONSIBLE INDIVIDUAL	22b. TELEPHONE (Include Area Code)	22c. OFFICE SYMBOL
Betty L. Fox	(202) 325-7042	DNA/STTI

DD FORM 1473, 84 MAR   83 APR edition may be used until exhausted.
All other editions are obsolete.

SUMMARY

Development of the Large Blast/Thermal Simulator requires the investigation of certain critical components to determine the feasibility of their design and use in the facility. One such group of components is the driver tube and devices used to release the charge pressure from the tube. Blast wave simulation is accomplished by the simultaneous rapid release of pressure from a number of these tubes mounted with their exit nozzles in the same plane. In order to accurately model the wave front produced in a nuclear blast the pressure release must be on the order of milliseconds. Two possible methods of pressure release considered for the LB/TS are explosively ruptured diaphragms and fast acting valves. This report focuses on a fast-acting sliding-sleeve type valve which has two distinct advantages over a diaphragm. The valve requires much less turnaround time between tests as opposed to a diaphragm and a rapidly closing valve can be used to better tailor the expansion phase of the blast wave for better blast simulation.

The study of a sliding sleeve valve for the LB/TS drew heavily on Sverdrup's previous experience with valves of a similar type which had been successfully fabricated and tested for other impulse facility applications. These valves were smaller in size and rated for lower pressure than the valves proposed for the LB/TS. The bulk of this study consisted of exploring the problems involved in scaling up these pilot HIRT valves to the size required for LB/TS use.

The feasibility study concentrated on determining the maximum size and pressure for which a valve of this type would be suitable assuming that ordinary fabrication methods and readily available hardware would be used. The study revealed that sleeve weight was a critical factor in achieving the basic requirement of an opening time of 50 milliseconds or less. Diameters in the range of 40-50 inches at pressures of 2000-3000 psi were found to be approximate upper limits if standard hardware items are used in the hydraulic system. An important part of the feasibility study was the identification of a line of fast-acting, high-flow, hydraulic valves which permit the required movement of sleeves in the 2000-lb. range.

The conceptual design focused on one valve size and pressure combination. A 36-inch diameter valve rated for 2500 psi was conceptually designed and a series of drawings was produced in sufficient detail to determine accurate costs. The two major areas of study in the conceptual design were hydraulic system and seal system performance. A hydraulic system was designed with sufficient flexibility that it could accelerate and decelerate the sleeve over a wide range of sleeve weights, friction loads, and differential pressure effects. The sealing system evaluation revealed some problems with differential contraction between the shell and sleeve, and problems in use of an elastomeric compound at the desired operating temperature of 650°F. These problems could best be resolved in a prototype development program.

A prototype development program is described in Section 3 of the report. Hardware items required and areas needing experimental study are discussed. Costs for such a program as well as costs for a prototype valve and production valves are given in this section.

In conclusion, use of a sliding sleeve valve, of the type studied, as a starting device for the LB/TS appears entirely feasible and would provide measurable savings in time and the ability to provide wave shaping which would not be possible using alternate methods such as diaphragms.

TABLE OF CONTENTS

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By ....

Distrib

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Spec...

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A-1

LIST OF FIGURES

LIST OF TABLES

SECTION 1
FEASIBILITY EVALUATION

1.1 INTRODUCTION

The purpose of the feasibility evaluation, the first phase of the study, is to examine two sliding sleeve valve sizes at various pressures to determine which valves, meeting the operating requirements, may be fabricated using conventional materials and methods and readily available, commercial hardware. The critical performance characteristic for all valves was an opening time of 50 msec or less. The specific charge pressure and size combinations were selected to give valve flow areas equivalent to diaphragms in the 25" to 40" diameter range. They include a 25"-I.D. valve at charge pressures of 2000 psi and 4000 psi and a 40"-I.D. valve at 1500 psi and 2500 psi. Figures 1 and 2 show results of the study for two valve sizes and the typical arrangement for all valves.

The design and fabrication details of the sliding sleeve valves for the pilot HIRT project--smaller scale valves (12" dia. and 16" dia.) which were fabricated and tested in an earlier Sverdrup development program--were studied initially in the evaluation. These valves were successfully operated hundreds of times at charge pressures up to several hundred psi and are good beginning models for the larger scale valves. These larger valves differ from the pilot HIRT valves in that they are externally rather than internally pressurized. This arrangement results in greater mass for the major components but has the distinct advantage of eliminating the "self-closing" effect of the pressure imbalance on the sleeve as the valve begins to open. This effect was noticed in the pilot HIRT valves especially at higher charge pressures as the pressure differential was approximately proportional to charge pressure. It is anticipated that there will be a beneficial "self-opening" effect present in the externally pressurized design but this is not considered in sizing the actuating system. The effect must be evaluated in sizing the deceleration system, however.

The pilot HIRT valves are now located at the University of Texas at Arlington, Aerospace Laboratory. A trip was made to U.T. to recover copies of drawings of the valves, verify hardware components
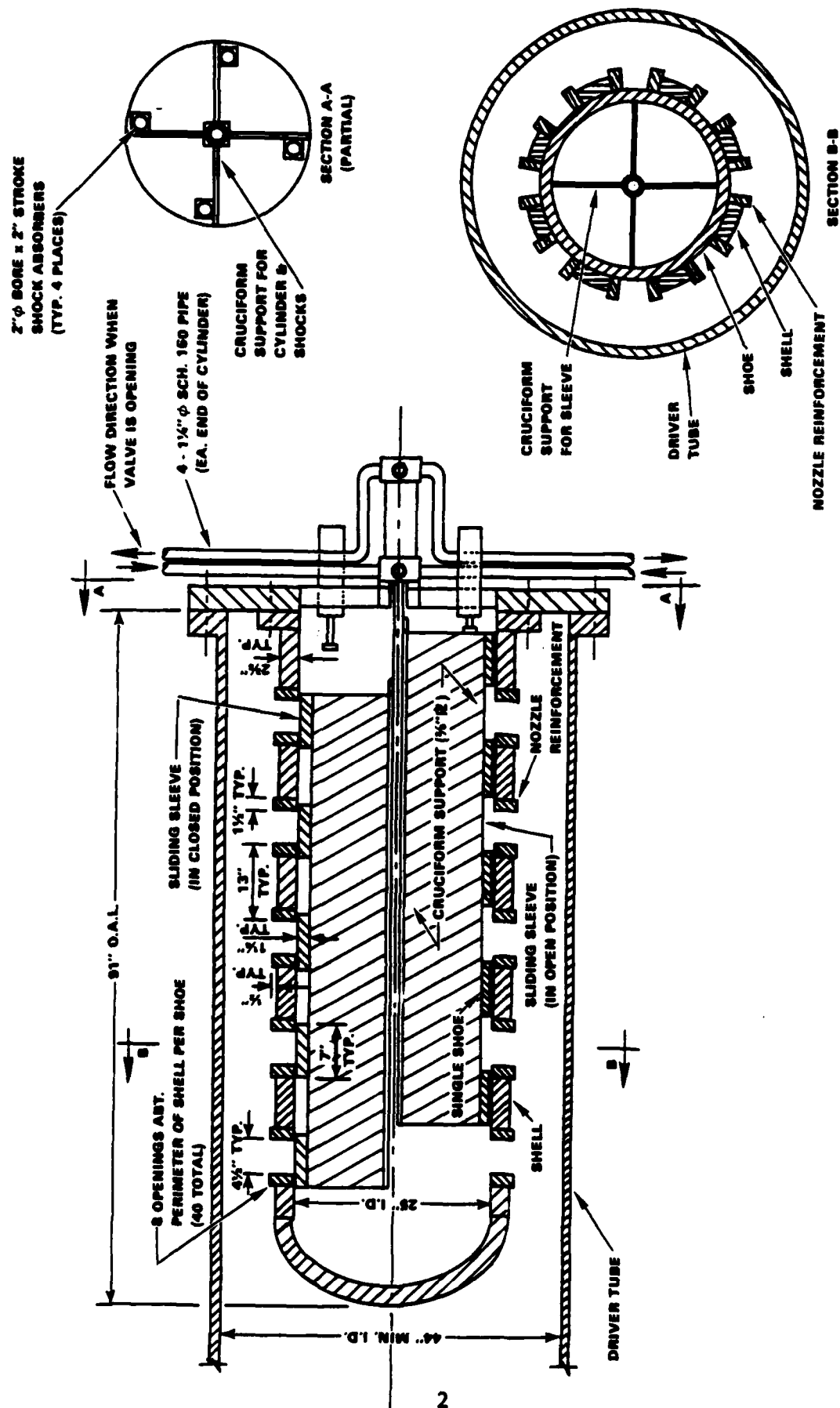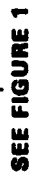
Figure 1. 25" φ Sliding sleeve valve, 4000 psi charge pressure.

2

Figure 2. 40" φ Sliding sleeve valve, 2500 psi charge pressure.

in use, and to talk to U.T. personnel about valve operation experience. The 12"-diameter valve is currently being used with the pilot HIRT tunnel under the direction of Aerospace Engineering Professor, Dr. D.R. Wilson. Although operational experience is limited, the sliding sleeve valve has operated well at low pressures with no significant maintenance problems.

The following is an explanation of valve operation and the function of major components of the valve as shown in Figures 1 and 2.

Shell

The shell carries the majority of the charge pressure load when the valve is closed. In this application the shell is externally pressurized and, therefore, buckling of the shell is the critical failure mode, resulting in shell thicknesses greater than required for an internally pressurized shell. The large number of openings in the shell reduces its strength and, in order to minimize the shell thickness required, each opening is reinforced by the installation of a forged steel nozzle. The summation of the cross-sectional areas of all openings must be greater than the flow area of the valve and, therefore, several annular bands of openings are required. By comparing Figures 1 and 2 it can be seen that the number of these annular bands increases with valve diameter producing longer valves for the larger diameters. The inner surface of the shell will be machined to provide a good seal and smooth sliding surface for the sleeve.

Sleeve

The sleeve consists of a number of individual rings or shoes which carry the charge pressure load from the openings in the shell. Each shoe is sealed by 0-rings at raised edges on each end of the shoe. These raised edges are the surfaces on which the shoe slides when the valve is opened. All the shoes are connected to a cruciform support consisting of four perpendicular arms constructed of 3/8" plate. The cruciform support is, in turn, connected to the hydraulic cylinder or actuator which rapidly shifts the sleeve to open the valve. As can be seen in the upper portion of Figure 1, when the valve is closed, each shoe seals one annular band of openings around the shell. The lower portion of the same figure shows that the shoe moves into the area between shell openings when the hydraulic cylinder is pressurized and the valve opens.

Drive System

The rapid shifting of the sleeve is essential to produce the desired valve performance. A valve opening time of 50 msec required an average velocity of approximately 10 feet per second. To produce this velocity, a high-pressure, high-flow rate drive system must be provided. This is accomplished with a single hydraulic cylinder as shown in Figures 1 and 2 and solenoid-actuated valving as shown in Figure 7. Sleeve weights in a range from 1000 lbs. to

4

3700 lbs. must be accelerated by this system. Sleeve deceleration is accomplished by flow metering through the blind end of the cylinder in conjunction with four shock absorbers shown in Figures 1 and 2.

In studying the 25"-I.D. and 40"-I.D. valves, each is broken down into its component parts and each part is evaluated independently to determine the effects of size and pressure on certain critical parameters such as weight of sleeve, O.D. of shell, flow rate to cylinder, etc. The preliminary sizing of each component is discussed in the following sections and potential problem areas are identified. The supporting calculations for these sections are filed in the Defense Nuclear Agency Technical Library.

1.2 SHELL DESIGN

Shell design consists of determining opening size and number, shell length, and shell thickness. Opening size is calculated based on the assumption that sleeve average velocities of the study valves will be approximately the same as sleeve average velocities of the pilot HIRT valves. The velocity obtained in the HIRT valves was 10 feet per second and this velocity was assumed for the sleeves of the study valves. Using this data and the conservative assumption that the lead distance of the shoe (overlap at front of opening) plus opening distance must be traversed in 50 msec led to an opening size of 4-1/2" diameter for all valves. The lead distance is the distance the sleeve must travel before the valve starts to open and for the study valves it is set at approximately 1-3/4". The number of openings required is based on data from the pilot HIRT valves showing that the summation of the areas of all openings must be 1.3 times the basic cross-sectional area of the valve to assure full flow. This requirement puts limitations on the diameter of valve possible with this design since the perimeter distance available for location of openings increases linearly with the diameter while the number of openings required increases with the square of the diameter. For large-diameter valves the sliding sleeve length (and, therefore, the weight) may become unacceptably large.

The standard used in calculating shell thickness and opening strength is the ASME Boiler Pressure Vessel Code, Section VIII, Division 2. To minimize thickness requirements a high-strength, heat-

treated steel (ASTM A-517 Gr. F) was assumed for shell fabrication. Figure 3 shows the relationship of shell O.D. to charge pressure for the basic diameter valves studied. As can be seen from the graph, as valve diameter grows larger, shell thickness (and, therefore, O.D.) must be greater for a given charge pressure. The rate of increase in shell thickness for increasing charge pressure is also somewhat greater for the larger diameter valves. The graph consists of three curves for each diameter for the purpose of showing the efficiency of reinforcing the shell openings. It can be seen that with nozzle reinforcement, as shown in Figures 1 and 2, shell O.D. can be kept close to that required for a shell with no openings. Shell O.D. is an important criterion from the standpoint of fabrication and material cost, but is equally important in determining the size of the driver tube into which the valve must be inserted. To ensure full flow, the external driver I.D. was sized such that the annular cross-sectional area around the valve shell was at least 1.5 times the area of the shell I.D. cross-sectional area.

Based on the study of shell requirements for stress it is concluded that the smaller diameters yield more efficient designs, but there should be no great difficulty in fabrication of any of the valve sizes studied here.

1.3 SLIDING SLEEVE DESIGN

The design of the sliding sleeve consists of determining individual "shoe" thickness and width as well as establishing cruciform-type plate support requirements. Refer to Figures 1 and 2 for sliding sleeve configuration.

Since a critical parameter in the design of the valve is the mass of the sliding sleeve, the same high-strength, heat-treated steel (ASTM A-517 Gr. F) assumed for the shell will be used. Each sleeve shoe thickness is computed by assuming that the tangential bending stress will not exceed 50% of the yield stress nor 50% of the elastic buckling stress. Yield and buckling stress levels in the shoe are determined assuming four different types of load-resisting mechanisms: ring yield action, ring shrink buckling, arch yield, and arch buckling between cruciform supports. The controlling mechanism for the final design will
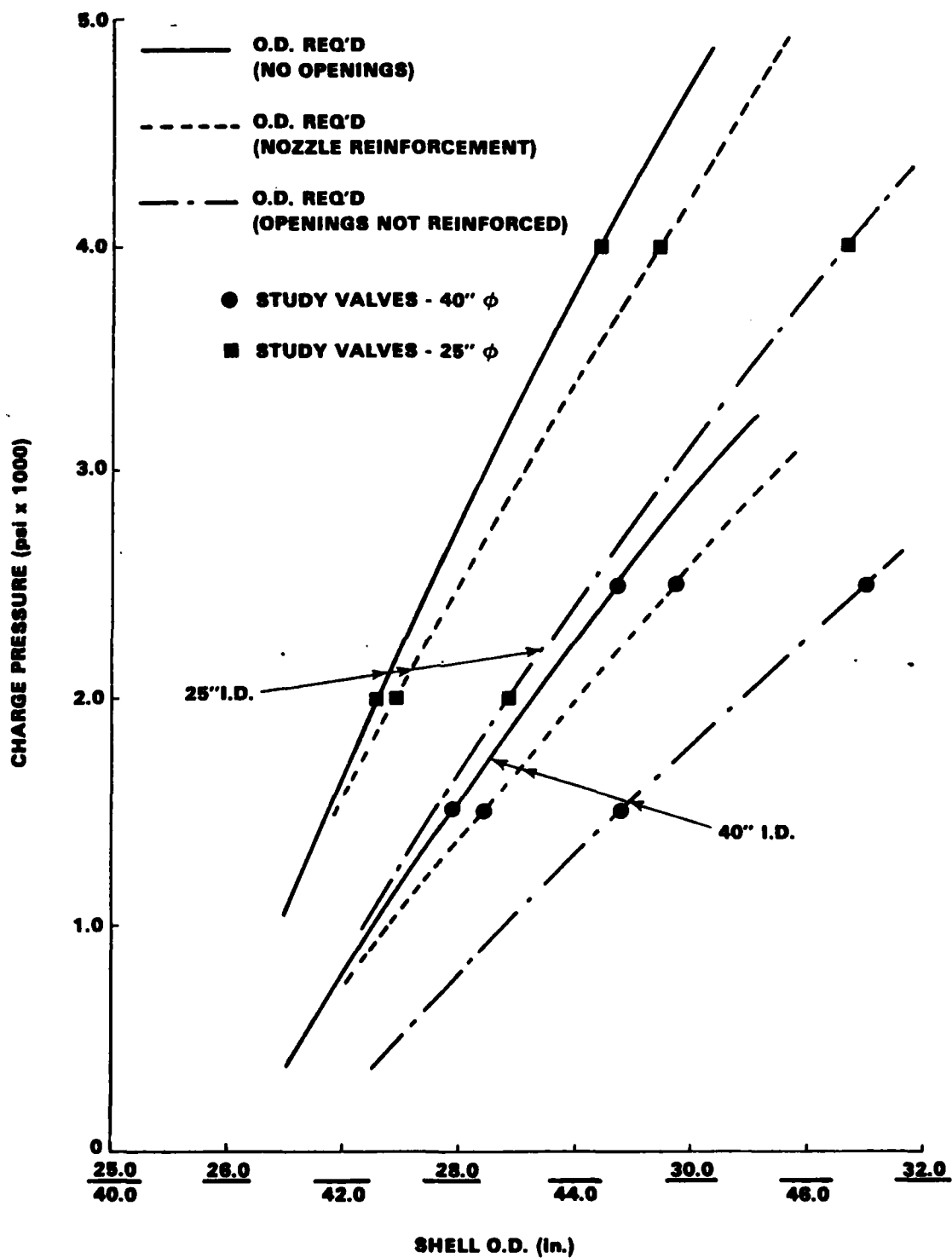
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Figure 3. Shell O.D. vs. charge pressure.

7

be a combination of ring yield and arch yield but for the purposes of the feasibility evaluation each mechanism can be treated separately. The controlling stresses for all shoe diameters and thicknesses under study here are developed by arch yield action.[1] Although arch yield controls for the shoe, compressive loads on the cruciform-type plate supports are very low for this case. The controlling stress in the cruciform plates is a shear stress developed when the valve is being opened. A plate of 3/8" thickness was conservatively assumed for all valve sizes.

The results of the sleeve sizing calculations may be seen in Figure 4 showing sleeve weight versus charge pressure. This graph illustrates the fact that as valve diameter increases for a given charge pressure, sleeve weight increases rapidly. The rate of increase in sleeve weight versus charge pressure is greater for the larger diameters also. This is illustrated by reference to Figures 1 and 2. In comparing the figures it can be seen that five shoes or rings are required for the 25"-diameter valve while eight shoes are required for the 40"-diameter valve causing a substantial weight increase. The findings. of this portion of the study are consistent with those of the shell design and indicate that, although all valve sizes studied can be fabricated, based on stress, the smaller diameters produce the most efficient valve designs.

1.4 HARDWARE SELECTION

As shown in Figures 1 and 2, a single double-acting hydraulic cylinder is used to rapidly drive the sleeve into the open position and is also used to close the valve. The selection of this cylinder and the control circuit valving required to actuate it is a crucial part of determining the feasibility of sliding sleeve valves of this type and size. Deceleration of the sleeve is an important factor as well since this must be accomplished in a short distance to minimize sleeve length and weight. Hardware items were selected from among standard components readily available from a number of manufacturers.

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[1] see Appendix
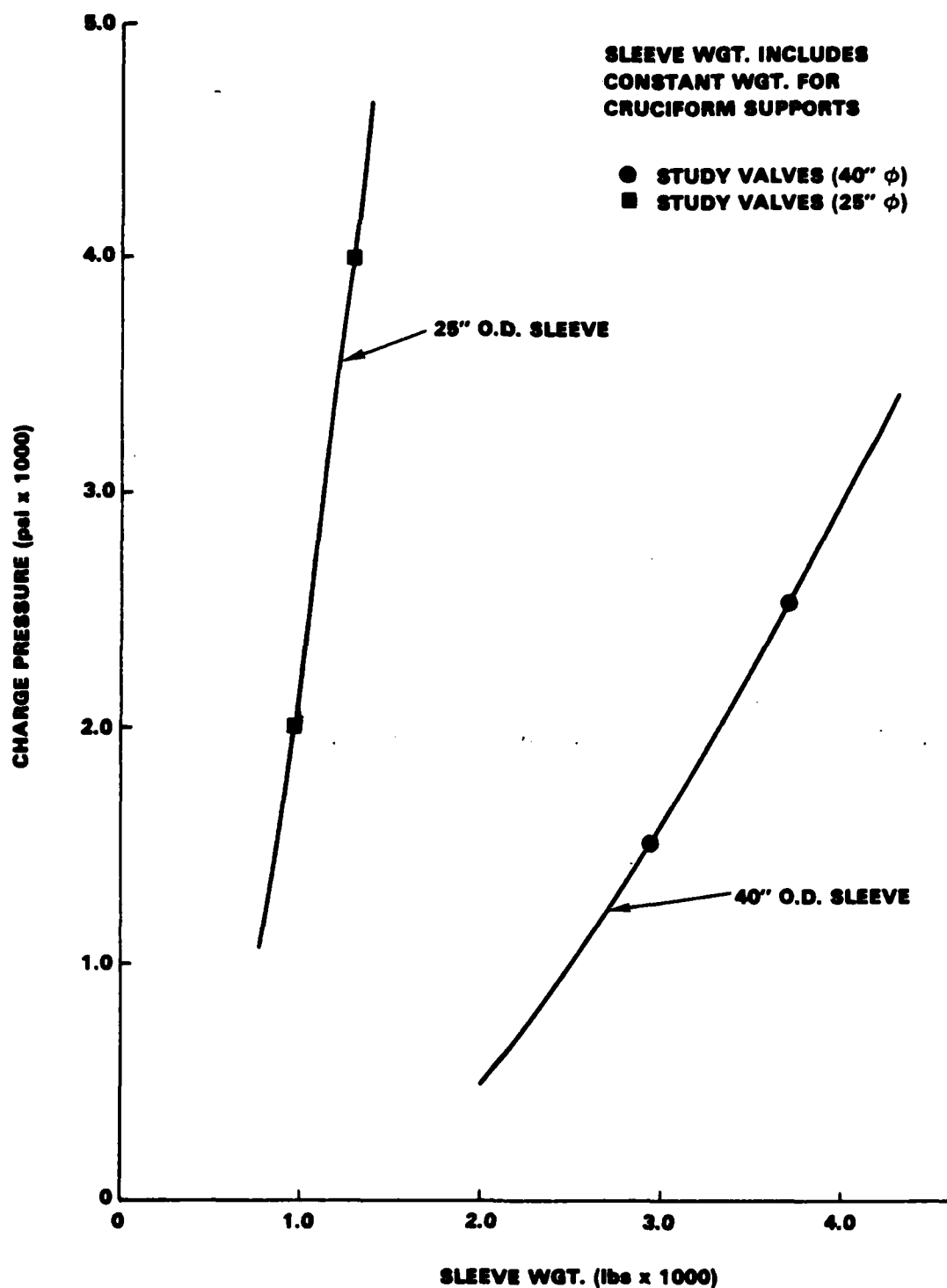
Figure 4. Sleeve wgt. vs. charge pressure.

9

The hydraulic cylinder is sized based on the requirement for a valve opening time of 50 msec. This opening time requires an acceleration of approximately 13 g's[1] for the shell opening size (4½") selected and, therefore, the cylinder has to supply a force equal to 13 times the sleeve weight plus a small friction force. Figure 5 shows the required cylinder area as a function of sleeve weight for three hydraulic pressures. Standard cylinder sizes are also shown to the left of the graph. As is shown, the larger sleeve weights require a hydraulic pressure of 3000 psi or greater to maintain a reasonable cylinder size.

In addition to the force requirement, the 50-msec opening time limitation requires that the flow of hydraulic fluid to the cylinder commence very rapidly and that the flow rate be quite high. Direct-acting solenoid valves from various manufacturers were studied for the purpose of controlling this flow, but at the required pressure the number of small orifice valves required for the flow rate is unreasonably high. A more efficient arrangement is the use of solenoid valves to actuate larger cartridge-type valves. Cartridge valves have opening times comparable to those for solenoid valves (<25 msec) and can provide flow rates sufficiently high such that only one or two are needed for each end of the cylinder. Four inlet and outlet lines are required at each end of the cylinder to provide the required flow and to line up with the cylinder cruciform support to minimize restriction of the air flow from the valve. Figure 6 shows the relationship of sleeve weight to flow rate to the cylinder. Shown to the left of the graph are desirable line sizes to and from the cylinder and required cartridge valve sizes. Figure 7 shows the flow control schematic indicating the function of the solenoids and cartridge valves.

Deceleration of the sleeve will be accomplished by a combination of flow metering through the hydraulic cylinder and possibly the cartridge valve in addition to the use of four short-stroke shock absorbers mounted as shown in Figures 1 and 2.

From the study of hardware requirements it has been established that adequate standard hardware is available from a number of manufacturers.

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[1] see Appendix
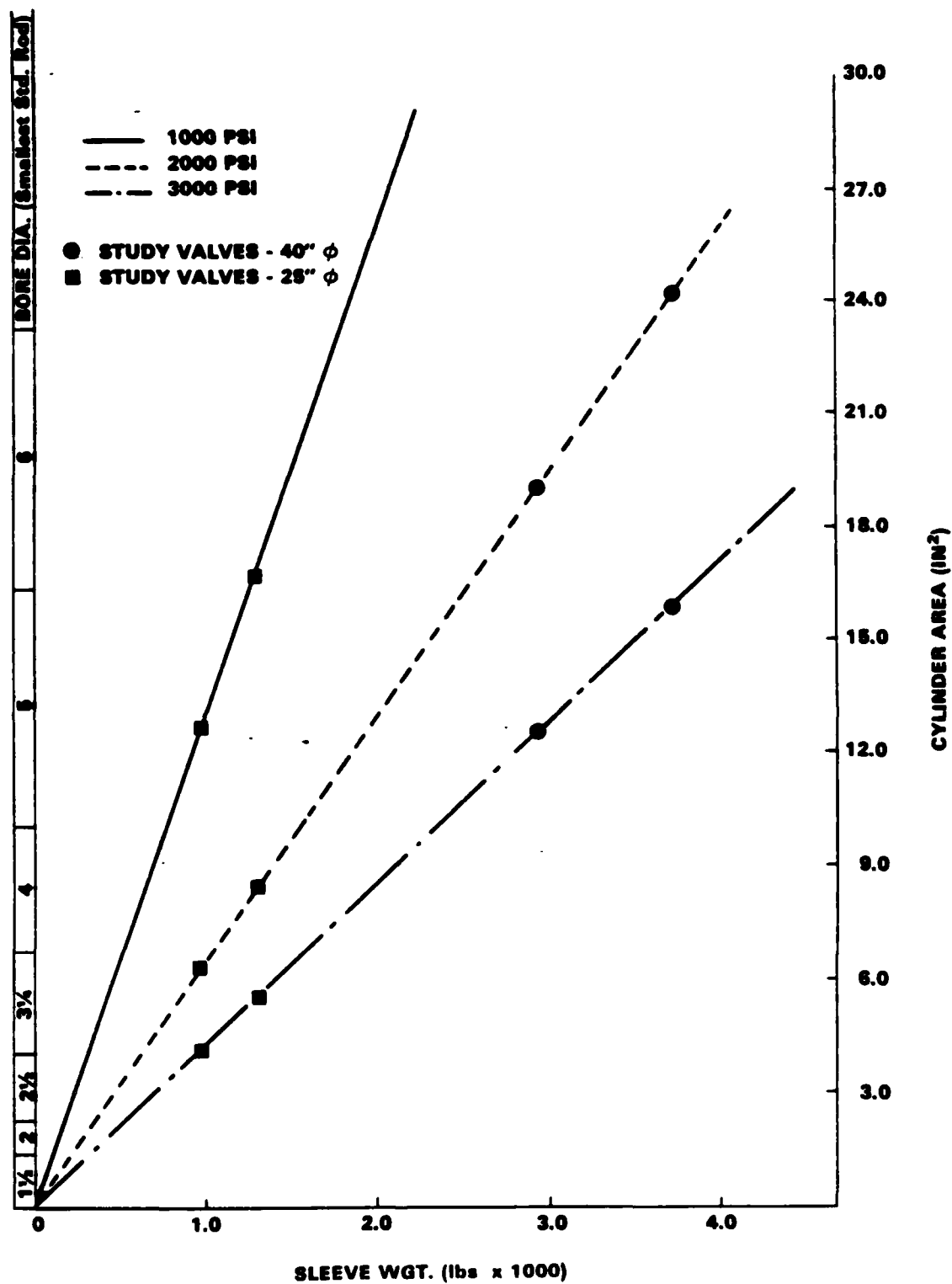
Figure 5. Sleeve wgt. vs. cylinder area.

11

Figure 6. Sleeve wgt. vs. flow rate (to cylinder).

12

HYDRAULIC CYLINDER

4 PORTS FOR EA. END OF CYLINDER

Ⓒ Ⓓ

A A

B B

Ⓐ Ⓑ

B A A X B

T P P T

PUMP

TANK

TANK

ALL ITEMS WITHIN BOUNDARY MOUNTED IN ONE ENCLOSURE

Ⓐ — 2 POSITION — 2 WAY SOLENOID (4 PORTS)

Ⓑ — 2 POSITION — 2 WAY SOLENOID (4 PORTS)

Ⓒ-Ⓓ CARTRIDGE VALVE   SYSTEM PRESSURE - 3000 psi

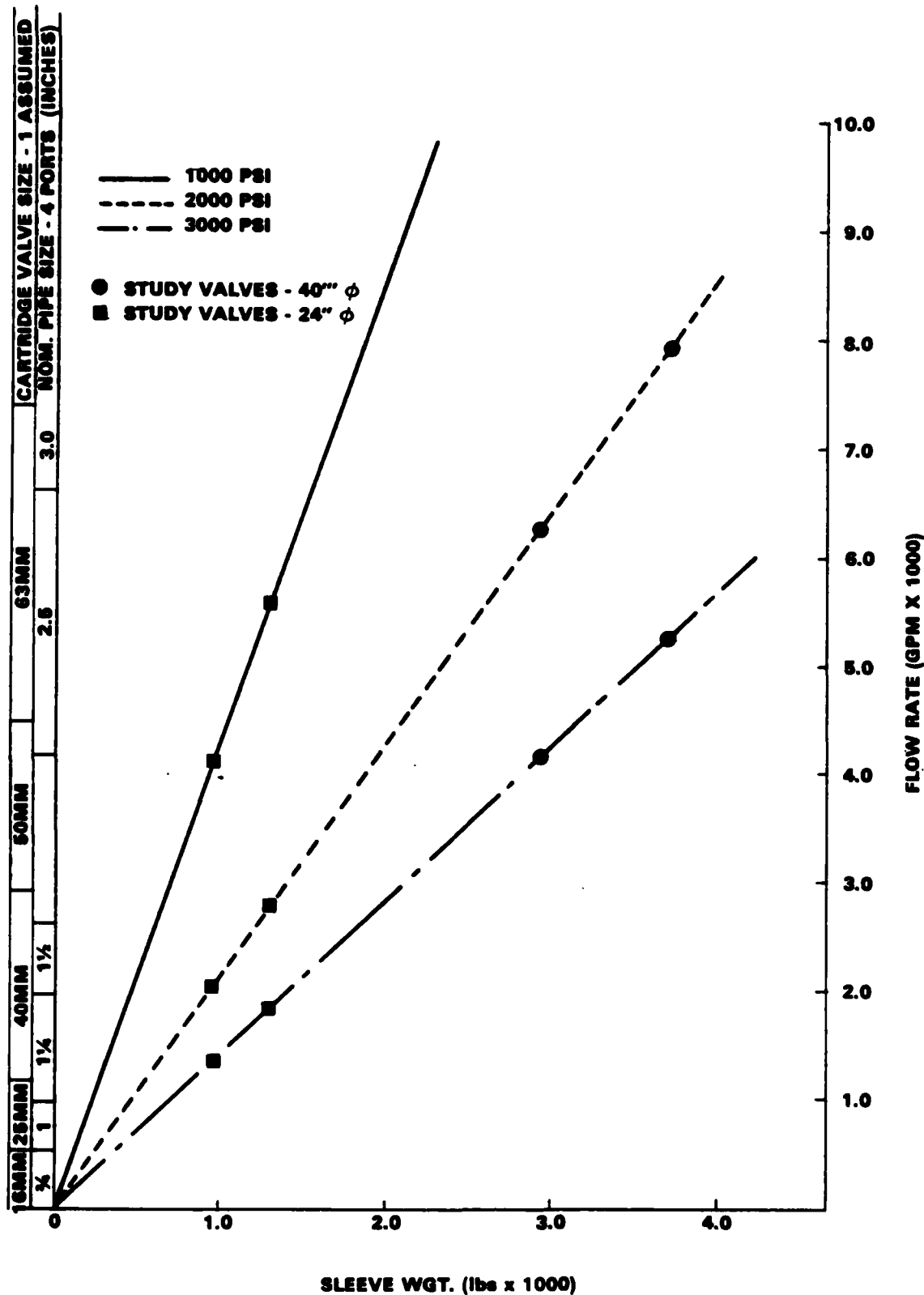**Figure 7. Flow control schematic for cylinder.**

1.5 CONCLUSIONS

In summary, sliding sleeve valves with size, performance, and pressure rating characteristics of the type studied can be produced using common methods of fabrication, materials, and hardware. The degree of difficulty in producing a sliding sleeve valve of this type increases with increasing diameter. Sleeve weight is the critical limiting factor and sleeve weights over 2000 lbs will pose difficulties. Diameters in the range of 40-50 inches at pressures of 2000-3000 psi appear to be approximate upper limits. Sleeve weights for larger valves would be on the order of several tons and would require exotic hardware to accelerate and decelerate them at the rates required.

SECTION 2
CONCEPTUAL DESIGN

2.1 INTRODUCTION

In the conceptual design phase the work of the feasibility evaluation is extended with emphasis on the design of a valve for one size and pressure combination. The selection of this combination is based on charge pressure optimization studies for the LB/TS conducted by DNA and the results of the feasibility evaluation portion of this report. Three valve combinations were generated by the LB/TS studies. In order of desirability they are: 50-inch diameter at 1500 psi, 40-inch diameter at 1500 psi, and 36-inch diameter at 3000 psi. The feasibility evaluation showed that the smaller valve sizes at lower pressures were easier to fabricate and offered a high probability of success. The conceptual design valve size will be the basis for the cost studies for the prototype valve and for the costs of the production valves. In view of the importance of providing a workable valve design as the model for estimating costs, the smallest valve size, 36-inch diameter, is selected for conceptual design. To further ensure a successful valve, a somewhat lower design charge pressure of 2500 psi is specified. The 36-inch valve size, for conceptual purposes, is defined as a nominal size. The actual physical diameter of the valve I.D. will be that which will provide a flow area equivalent to a clear 36-inch diameter opening after all blockages such as cylinder, supports, shocks, etc., are accounted for.

Features for study, which were not considered in the feasibility evaluation, include seal performance, machining tolerances, dimensional changes due to pressure, friction forces, plating requirements, the effects of temperature, and details of the cylinder support and flange at the valve outlet. The feasibility phase assumptions for details of attachment of the valve to driver tube proved to be too inefficient at the conceptual design valve size and pressure. A conical transition is added to the driver tube to maintain reasonable flange thicknesses. See Section 2.2. Differential compression of the shell and sleeve due to pressure is investigated for its effect on seal performance in

Section 2.4. The effects of a design temperature of 650°F have been accounted for in the design of the steel portions of the shell and sleeve in both the feasibility and conceptual phases. In the design of an elastomeric O-ring seal, temperatures in this range are a critical factor. The discussion of this item appears in Section 2.4.

In order to better predict the performance characteristics of the valve, a hydraulic circuit is designed in detail with sizing of all major components of the system. The design is based on the type of system required to actuate one valve (such as would be used for prototype testing) but the techniques used could be scaled up to operate multiple valves. Several options are possible for the circuit but the one chosen for the conceptual design is a balanced accumulator approach. It is discussed in detail in Section 2.5. A set of pressure and sleeve position diagrams with respect to time is also presented in this section.

The following drawings have been produced and are referred to throughout the conceptual design portion of the report. They show details of the valve construction as well as a prototype test set-up and a possible LB/TS arrangement of 18 production valves.

Drawing No.	Title
9048-M-1	Test Arrangement and Installation
9048-M-2	Assembly (Sheet #1)
9048-M-3	Assembly (Sheet #2)
9048-M-4	Sleeve Assembly

2.2 SHELL DESIGN

The conceptual design of the shell is an extension of the design work done in the feasibility evaluation. It requires material selection, a stress analysis, determination of opening size and location, and establishment of machining, plating, and other fabrication detail requirements. Guidelines of the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, are again used in the calculation of wall and flange thicknesses and in establishing welding requirements.

The nominal valve size under consideration is 36 inches in diameter. The definition of nominal size as given in the introduction (Section 2.1) requires a free flow area equivalent to a clear opening of a diameter equal to the nominal size. For a 36-inch diameter this area is 1018 square inches. In order to achieve this area, considering the blocking effects of the hydraulic cylinder, shock absorbers, sleeve, etc., a shell of approximately 48 inches in diameter is required. The inlet ports around the shell remained 4.5 inches in diameter as determined in the feasibility evaluation based on expected drive system performance. A total opening area of 1.3 times the equivalent flow area of 1018 square inches is the basis for calculation of the required number of openings. The total of 85 openings so calculated is arranged in 5 bands of 17 openings each. The spacing between opening bands along the shell is increased from previous calculations to 11.25 inches to provide additional deceleration distance. The deceleration distance provided in the feasibility evaluation, 2 inches, resulted in decelerations of 41 g's which could produce warping in the lightweight sleeve. The acceleration and deceleration distances are made approximately equal to better balance the associated sleeve forces.

Material selection is unchanged from the feasibility phase, the high-pressure load and large valve diameter requiring a high-strength, heat-treated steel (ASTM A-517). At the diameter selected and the 3.25-inch thickness calculated, the shell can be fabricated by rolling, eliminating expensive forging methods. Forged steel reinforcing rings of the same material will be used at the ports to maintain a minimum wall thickness.

At the outlet of the valve the original intent was to provide a bolted connection from the valve flange to an adapter flange which would in turn bolt to a flange at the end of the driver tube. (see Figure 2). The thickness required for the adapter flange in this arrangement at 2500 psi approaches one foot, therefore, in order to reduce the eccentricity of this joint, a conical section is added to the driver tube. A bolted flange connection was evaluated for attachment of the valve to the conical driver end but, again, the necessity of allowing sufficient space for the bolts and nuts produces an eccentricity which requires a very thick flange. A welded connection provides the

17

minimum eccentricity possible and will be used at the valve-driver tube intersection. A bolted flange connection will be used in the straight section of the driver tube for valve attachment. (See Drawing #M-3.)

After rough fabrication of the shell including the completion of all welding operations and final heat treating, the inside or bore diameter of the shell will be hard chrome plated and the surface will be machined to a 16-micro-inch finish. This technique was successfully used in the fabrication of the 16-inch-diameter pilot HIRT valves and is recommended for dynamic O-ring seal applications. (See seal design, Section 2.4). The temperature limit for the chrome plating is approximately 500°F.

The cruciform support system for the hydraulic cylinder must resist the loads associated with acceleration and deceleration of the sleeve as well as dynamic pressures generated when the valve opens and gravity loads. Cruciform leg thicknesses should be minimized to reduce blocking effects but mild steel (ASTM A-36) can be used to provide thicknesses of 1.25 inches or less. A box fabricated from plate is used at the intersection point of the cruciform members in order to transfer the moments and shears from member to member and yet allow for rod passage from cylinder to sleeve. Box dimensions are based on stress requirements and the dimensions of a standard cylinder mounting plate. The attachment of the cruciform legs at the shell is designed as a bolted connection accessible from the exterior of the valve for ease of assembly and disassembly for sleeve removal and seal maintenance. The method of attachment would also permit the future use of Belleville washers if desired to provide additional deceleration distance. The Belleville washers would be placed on the bolts securing the cruciform brackets (between the bolt head and bracket) as a safeguard against overstressing the cruciform supports on deceleration of the sleeve. They would allow the sleeve to travel beyond the normal decelerate distance to a position partially blocking the next line of ports and would therefore be used at the early portion of the prototype stage of development and abandoned for final design. (See Drawing #M-2.)

Sufficient details were established in the conceptual design that accurate costs can be determined.

2.3 SLIDING SLEEVE DESIGN

Sliding sleeve design in the conceptual stage consists of more detailed stress analysis of the major components, investigation of pressure effects on sleeve dimensions for the purpose of determining clearances, groove detailing, and finish requirements. O-ring seals are discussed in Section 2.4. The feasibility evaluation assumption of an individual shoe width equal to 7 inches is reduced to 6 inches to provide a greater deceleration distance and minimize sleeve weight while keeping sleeve length a minimum. The lead distance, or overlap, of the shoe at the upstream end of the port does not provide sufficient benefit (in allowing acceleration in advance of opening) to offset the weight penalty caused by a wider shoe. (See Drawing #M-4.)

Stress calculations were carried out in much the same way as the feasibility evaluation by conservatively assuming that arch yield action controls for stresses in the shoe. Corrections for shear and axial deformation were made in an effort to provide more accuracy in the calculation. A total shoe thickness of 1.65 inches was determined by this method giving a sliding sleeve weight, including cruciform and moving parts of the cylinder, of 2450 pounds. Allowable stress levels are predicated on provision of a safety factor of two against yield or buckling. This is a lower safety factor than that provided by the ASME Boiler-Pressure Vessel Code which is used for the design of the shell. Justification for the decrease in safety factor is based on the consequences of a ductile sleeve failure. This failure would result in a rapid decompression when the shoe deflected enough to break the O-ring seal. This is not a catastrophic failure mode such as would result if the shell were to buckle. Of course it is advantageous with respect to valve opening time to use the lowest practical safety factor for the sleeve to minimize weight. (See Drawing #M-4.)

The most serious problem which results from the use of different safety factors for the shell and sleeve is the differential compression that results under pressure loading. The pressure loading on the relatively thin sleeve causes it to contract a greater amount than results from basically the same pressure loading on the thicker shell. This differential produces clearances in excess of those existing in the zero-pressure case and makes providing a seal more difficult. The

19

clearance increase beyond the machined original clearance is expected to be on the order of 0.011 inches, based on the safety factors currently being used. With more accurate stress analysis, more rigid inspection and testing, and more confidence in the design pressure (e.g. redundant pressure relief systems are provided, etc.), as could be provided for in the final design, safety factors for the shell could be reduced to permit better agreement between shell and sleeve deformation. See seal design (Section 2.4) for more details of the seal investigation.

The dimensions of the seal groove are taken from the design for the pilot HIRT valves and have proved successful in a dynamic application similar to that involved here. A brass overlay will be applied at contact points on the sleeve identical to that used in the HIRT valves to minimize scoring of the hard chrome surface of the bore. Scoring must be minimized to maintain seal effectiveness.

Cruciform support plates are designed for the shear load that results from acceleration and deceleration of the sleeve. The connection of sleeve and cylinder rod is designed for this loading condition as well.

The conceptual design of the sleeve, relying heavily on the successful pilot HIRT sleeve, allowed better definition of details which will result in simple calculation of a reliable cost.

2.4 SEAL DESIGN AND TEMPERATURE EFFECTS

The design of an adequate seal for the sliding sleeve is a critical part of the valve design. The method used must provide a static piston seal at 2500 psi across clearance gaps of 0.011 inches or more and the seal must be able to rapidly traverse the port opening without being damaged. It must resist blowout due to trapped gases during rapid decompression and have sufficient durability to operate for many cycles before time-consuming seal replacement is required. The seal design for the pilot HIRT valves successfully met similar criteria at somewhat lower pressures (1000 psi max.). For this reason an identical seal design will be used in this application.

20

A standard O-ring cross-section (0.210 inch diameter) will be installed in a special dovetail groove. The space remaining in the groove will be completely filled with a flexible sealant to avoid any cavities in which pressure could be trapped. Any pressure so trapped would cause an O-ring blowout when it crossed the upstream edge of the port opening and the O-ring would be cut as it crossed the down-stream edge.

An ordinary O-ring and groove design can seal a relatively large gap between the sleeve and bore due to the fact that the O-ring is compressed, by fluid pressure, in its groove over the full width of the ring. This allows high sealing pressures to be developed at the sur-face to be sealed. In the design of the seal for this application only the small portion of the O-ring above the groove is fully effective in deforming to produce a seal, therefore the clearances that can be successfully sealed by this method are much smaller than those for a typical O-ring. Further complicating the problem is the fact that under pressure the sleeve shoes contract more than the shell and clearances increase beyond those in the zero-pressure condition. The sleeve shoes are designed with a lower factor of safety and are therefore less stiff than the shell. (See sliding sleeve design, Section 2.3). Zero-pressure clearances must be provided such that the O-rings will be compressed or "squeezed" as much as possible without damage in the sliding or dynamic condition and yet have sufficient squeeze in the pressure-loaded case that a static seal can be provided. Of course the zero-pressure clearances must also be adequate for easy movement of the sliding sleeve. The fit specified on the conceptual drawings pro-vides a minimum clearance for sliding action. This fit will produce a gap that can be readily sealed by the O-ring. The final clearance for use in the production valve will be empirically determined during devel-opment of the prototype valve to produce the best seal and optimum clearance for sliding of the sleeve.

Selection of an O-ring compound must take into account factors such as possible reaction with the fluid to be sealed, abrasion resis-tance, amount of "squeeze" required to seal, seal permeability, resis-tance to environmental degradation, permanent set characteristics, and cost, among others. A pneumatic seal at high pressure is one of the

21

most difficult seals to produce because of several factors including difficulty in lubrication, excessive oxygen due to air compression (which causes more rapid deterioration of the seal), permeability (which produces some unavoidable leakage), and rapid decompression. Rapid decompression causes serious problems in permeable compounds because gases trapped within the material, when suddenly released, can cause the ring to blister or even rupture. The pilot HIRT valves utilized two different compounds for O-ring seals, a nitrile compound (Buna N®) and neoprene. The nitrile compound will be specified in this case due to its high abrasion and compression set resistance as well as its superior ability to resist rapid decompression. The temperature rating for this compound in prolonged dynamic service is 300°F.

The friction load which must be overcome in accelerating the sleeve is affected by many factors, the most important of which are the amount of "squeeze," the surface finish of the shell bore, the durometer rating of the elastomer, and the speed of motion. Friction calculations are conservatively based on 20-percent squeeze. The amount of squeeze at breakout with the sleeve under pressure will be considerably less. The surface finish of 16 micro-inches on the bore should provide sufficient smoothness for a low-friction load yet produce small cavities in the surface where lubricant can collect and further reduce the friction. Silicone lubricants specifically designed for O-ring use such as Parker "Super-O-Lube"® will be used to reduce friction. It is a long-lasting high-viscosity oil with a maximum temperature rating of 400°F. The durometer rating of the compound will necessarily be high (80°) in order to achieve a good seal and maintain abrasion resistance. This will unfortunately increase the friction load. The velocity of the sleeve will approach a peak of 18 feet per second at the point where deceleration begins, however, at the beginning of the stroke, velocities will be low and friction loads high. The total sliding friction (conservatively assuming full pressure acts on the seal throughout the stroke) is calculated to be under 4000 pounds. The static or breakout friction is expected to be from two to three times higher than sliding friction. High breakout friction is helpful in achieving the desired valve opening time, in that pressure is allowed to build in the cylinder before the

sleeve begins to move. Once sleeve movement begins there already exists an accelerating force two to three times higher than required to overcome sliding friction.

All elastomeric compounds are temperature sensitive. No compound currently available is rated for the specified operating temperature of 650°F under dynamic service. At this elevated temperature the volatile components of most elastomers are quickly driven off (the seal chars and becomes brittle) with the result that the seal loses its effectiveness and its resistance to abrasion. Likewise most lubricants are partially vaporized at this temperature leaving a sticky residue that further promotes seal failure. The compounds which provide the highest resistance to temperature are the silicones. They are rated to 450°F for continuous service. Unfortunately the silicone compounds have very low resistance to abrasion and tearing and therefore have poor service life in dynamic applications. Fluorocarbon compounds are rated at 400°F and are suitable for dynamic applications. The abrasion resistance of these compounds is much greater than that of silicone elastomers. The determination of the highest allowable design temperature which will permit a serviceable seal is best accomplished empirically at the prototype stage; however, without testing, it can be stated that a maximum operating temperature of 400°F could be achieved for many cycles of dynamic service with readily available compounds.

Another approach to providing operating temperatures of up to 650°F is to isolate the seal from the temperature except possibly for a brief period of time when the sliding sleeve valve is opened. One way this could be accomplished would be to use a flow-through heater, such as a pebble bed type, downstream of the valve. A thermal barrier could also be used as illustrated in Figure 8. The barrier would act only to prevent the movement of hot air to the area around the valve but would not take a pressure differential. It could therefore be made very thin. Moments before the sliding sleeve is to be fired, the shield would be hydraulically retracted to a position that will not restrict flow. The limited exposure of the seal to the high temperature flow would allow use of seal material such as nitrile with a temperature rating of 300°F. The use of cooling water coils is a feature, the need for which, would be determined by a heat transfer calculation if this approach were selected.
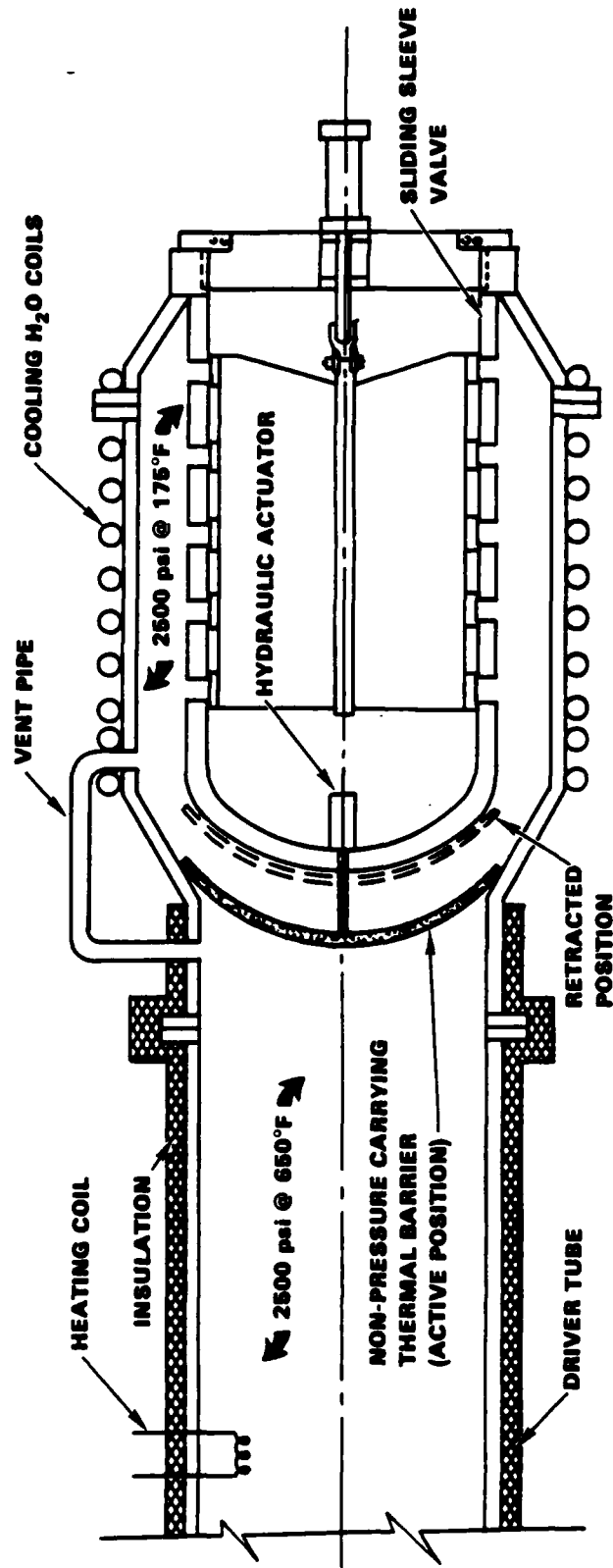
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Figure 8. Thermal barrier system.

24

In conclusion, the use of an elastomeric O-ring seal presents no major problems at temperatures up to 400°F. At 650°F the use of an elastomer would appear unlikely and provisions for isolating the seal would be required. Recent studies sponsored by DNA have shown that the use of valve closing for wave shaping may eliminate the need for heating the driver gas and the attendant high-temperature problems.

2.5 DRIVE SYSTEM DESIGN AND PERFORMANCE CHARACTERISTICS

The conceptual design of the drive system involves the selection and sizing of hydraulic circuit components and the calculation of approximate valve opening time and related performance characteristics. In the feasibility portion of the report, hardware items were selected based on the flow rates and pressures required to accelerate various sleeve weights to obtain the desired valve opening time of 50 milliseconds. In the conceptual design, efforts are concentrated on design of a complete hydraulic circuit to actuate a single sleeve weight of 2450 pounds.

The 50-millisecond opening time is so rapid that use of a single pressure compensating pump or series of pumps is not practical. A charged accumulator system is used in the hydraulic circuit to supply the sliding sleeve actuator. The circuit is shown in Figure 9. (See also Drawing #M-1.) It consists of two accumulators, one for acceleration of the sleeve, labeled "A", and one for sleeve deceleration, labeled "D", along with associated valving and pressure reservoirs to control flow direction and velocity. A list of hardware items is shown in Table 1. The circuit schematic is shown in the condition required for the initiation of the valve actuator cycle. At this point the sliding sleeve is in the closed position. Accumulator "A" is precharged with nitrogen (283 cubic inches at 3465 psia) as is accumulator "D" (378 cubic inches at 265 psia). The two pressure reservoirs, 6-I and 6-II (400-cubic-inch volume/reservoir), are precharged with nitrogen to the pressures required to affect the desired mid-stroke pressure adjustment in accumulators "A" and "D". The pressure reservoir, 6-III, used for controlled closing of the valve, will be charged to a suitable pressure.

Table 1. Hydraulic circuit components.

1. Hydraulic cylinder - 5-inch diameter bore, 11.25-inch stroke, 2-1/2-inch rod, [4] 2-inch ports each end
   Hydraulic Series N-7 or equivalent

2. Main valve - [2] 63mm cartridge valves (Vickers Series CVC) with manifold and pilot valve - 5000 psi

3. Acceleration accumulator - 600 cu. in. volume (total) 4000 psi

4. Deceleration accumulator - same as #3

5. Valve - 2-way pneumatic, 1 inch, 4000 psi Marotta Series MV167 or equivalent, [4] required

6. Pressure tank - 400 cu. in. - 1-inch connection, 3000 psi

7. Relief valve - 1 inch, 3000 psi

8. Silencer-Muffler - 1 inch

9. Check valve - 1 inch, 3000 psi, [2] required

10. Manual shut-off valve - 1 inch, 3000 psi, [3] required

To initiate the actuator cycle, the valve, "2", is shifted to the open position to allow flow to the cylinder. Calculations indicate that the sliding sleeve will reach mid-stroke (full open) within the required 50 milliseconds. At this point the valves 5-I and 5-II will be shifted to the open position. This will allow flow of nitrogen from accumulator "A" to the pressure reservoir 6-I to reduce the accumulator pressure to approximately 2000 psi and flow from 6-I to accumulator "D" to increase the accumulator pressure to approximately 1100 psi. This will affect a force reversal on the cylinder piston and provide the necessary deceleration force to stop the sliding sleeve at or before the end of its stroke. In order to hydraulically lock the sleeve in the open position, valve "2" will be closed at the end of the sleeve stroke.

The sliding sleeve can be moved to the closed position in either of two ways. If the speed of closure is not critical, the sleeve can be shifted by opening valve "2" and valve 5-IV to bleed the nitrogen pressure in accumulator "A" down to 200-300 psi. This will allow the pressure in accumulator "D" to shift the sleeve. If the speed of closure must be quite rapid, it will be necessary to also open

valve 5-III, in addition to the steps required above, in order to increase the pressure in accumulator "D". It may be necessary to add additional control circuitry to accumulator "A" to provide the deceleration force necessary to stop the sleeve as it approaches the closed position.

All of the directional control valves shown in the circuit are operated by pilot pressure controlled by solenoid valves. The pilot circuitry is not shown but will be required in the final design. Linear hydraulic dampers will also be used to provide deceleration forces in the last six inches of both the opening and closing strokes of the sleeve. Damper forces are not included in the calculations for sleeve movement but will not affect the opening time of the valve.

The hydraulic circuitry shown is that which would be provided for a prototype valve. The comparatively complex circuitry will be needed for testing a prototype in which critical parameters of the system, such as friction forces, pressure drops, pressure differentials, etc., can only be estimated. Empirically derived values for these parameters may permit the elimination of some of the circuit components. It is possible that the valve could be satisfactorily operated with accumulator "A" and two or three directional valves if the speed of valve closure is not critical.

The calculation of performance characteristics is illustrated graphically in Figures 10 through 13. In Figure 10, the change in accumulator pressures with respect to time is illustrated. For simplicity, the opening times for the valves controlling the flow is assumed to be zero and the time required for pressure adjustment between the accumulators and pressure reservoirs is assumed to be approximately four milliseconds. Although difficult to see, the pressure adjustment at approximately 46 milliseconds (or the point at which the sliding sleeve is full open) results in a force reversal on the sleeve at 50 milliseconds due to the larger piston area on which the deceleration accumulator pressure acts.

In Figure 11, sleeve acceleration with respect to time is calculated based on the accumulator pressure curves, taking into account pressure drop in the circuitry and friction load from the O-ring seals. Pressure drop is assumed to vary directly with velocity and friction load is
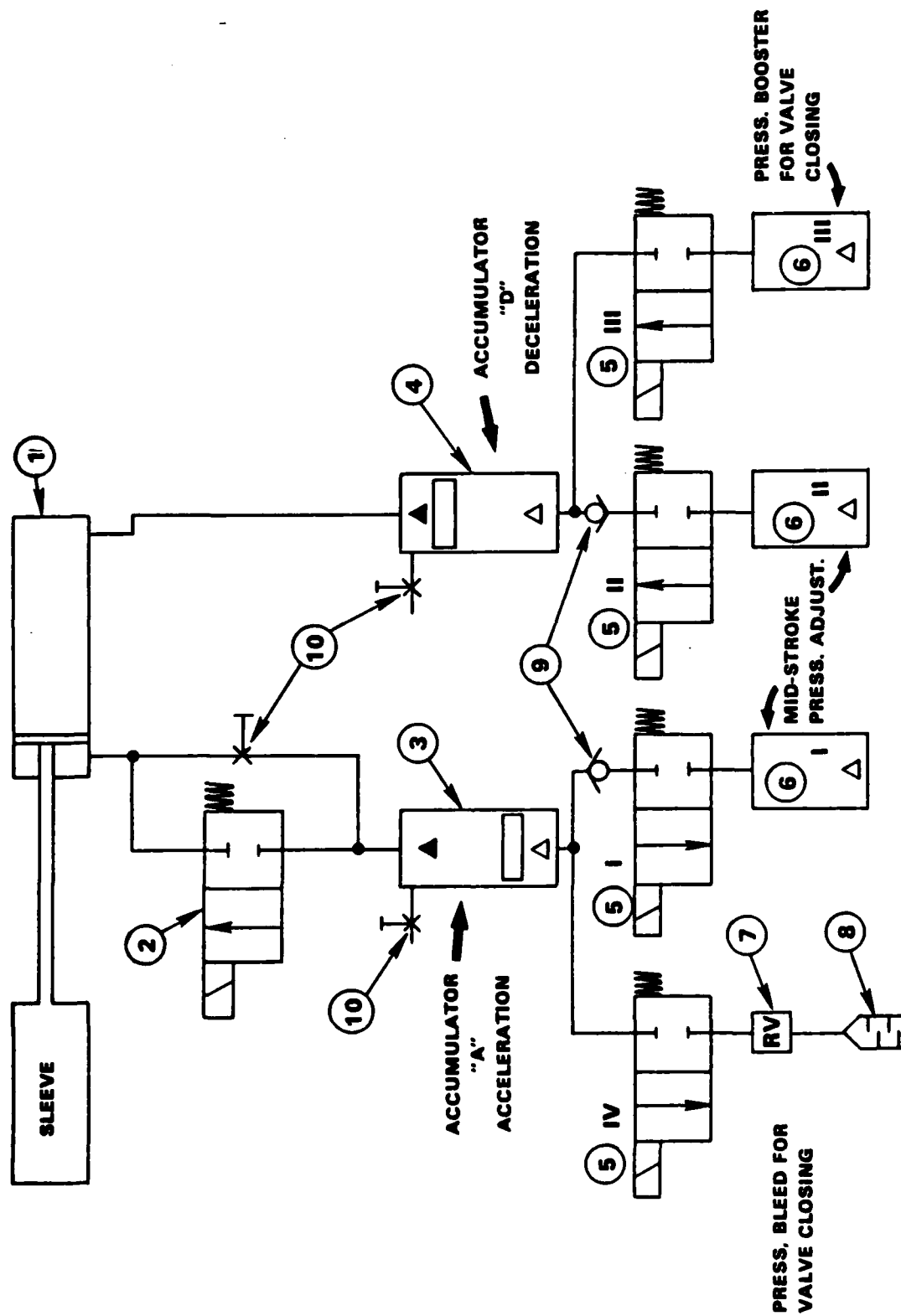
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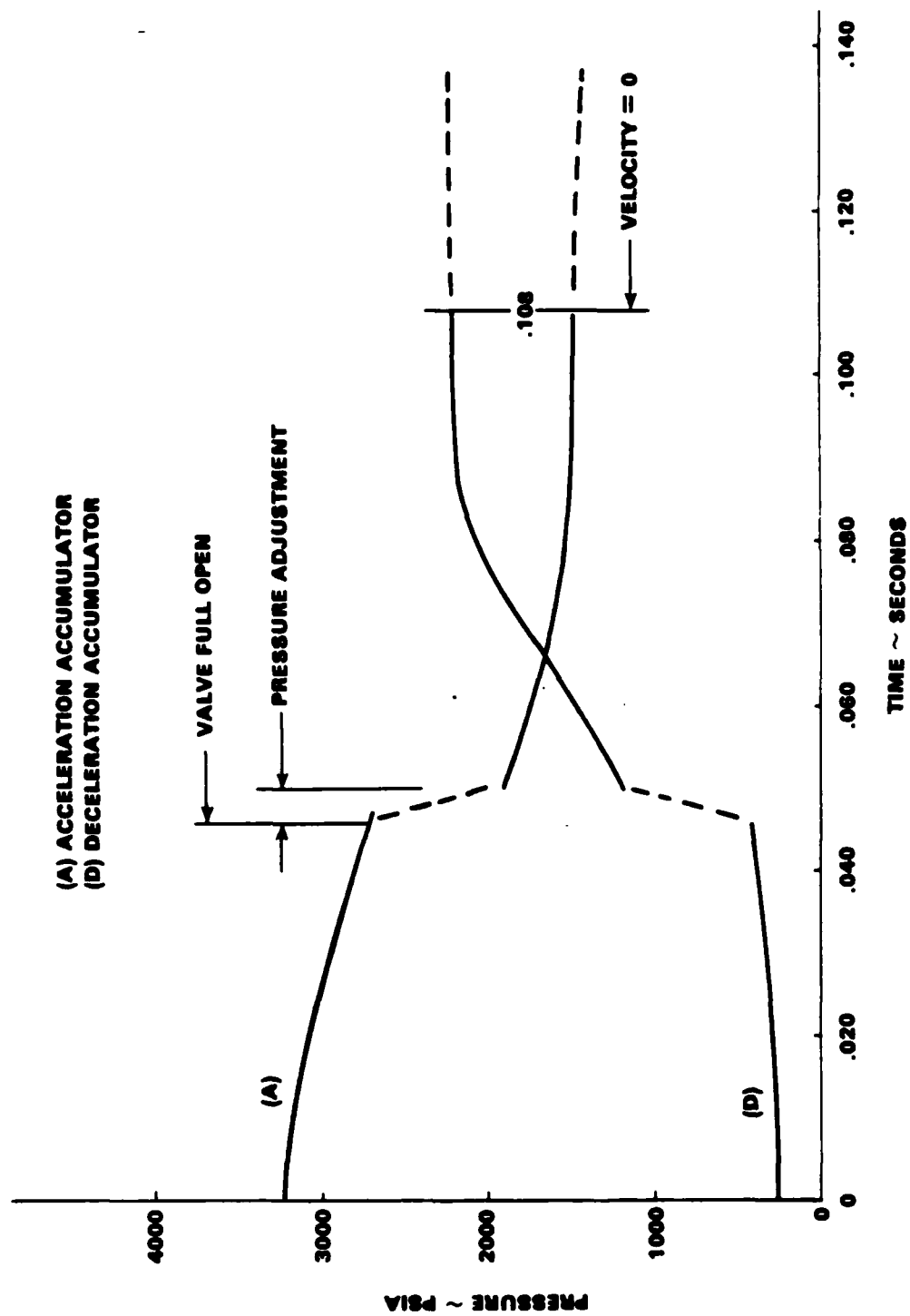Figure 9. Hydraulic circuit for actuator.
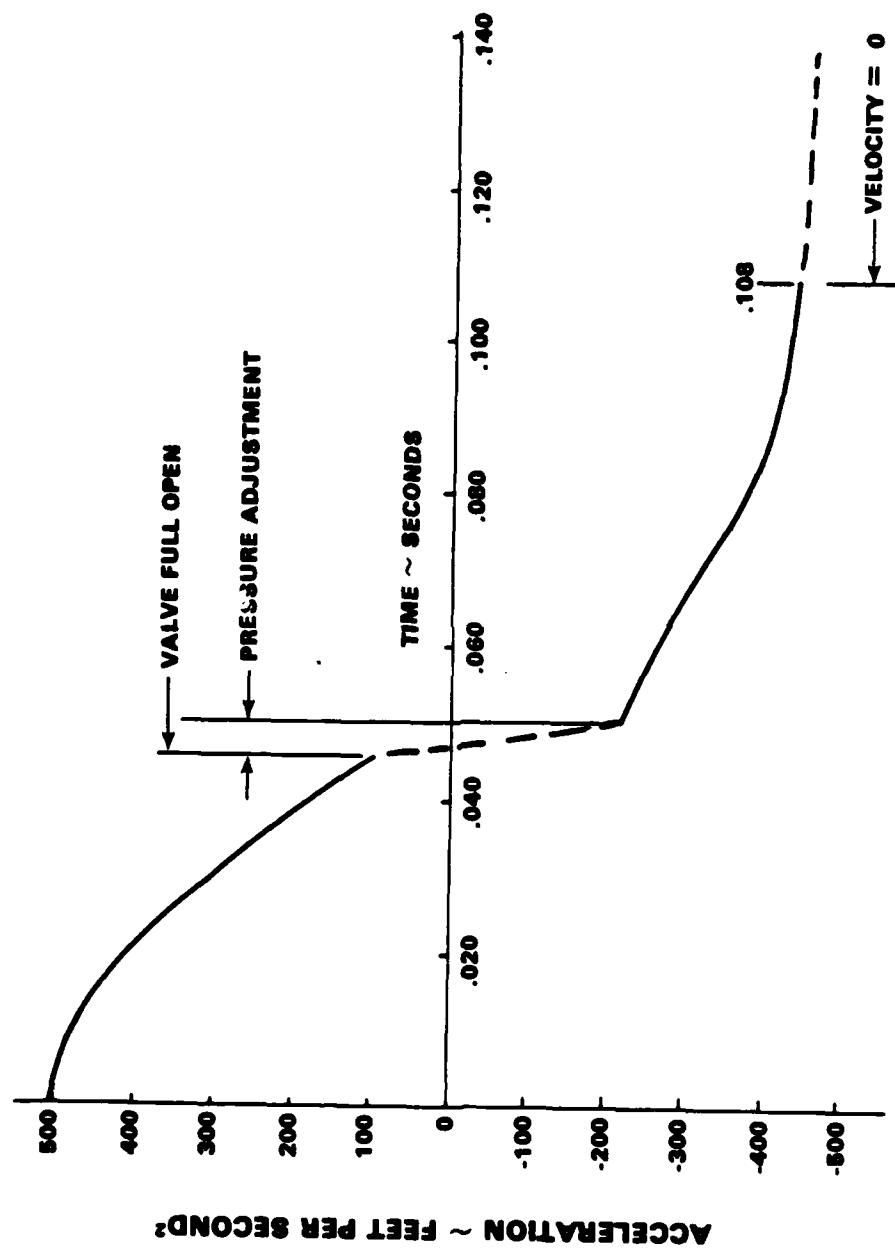
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Figure 10. Accumulator pressure.

29

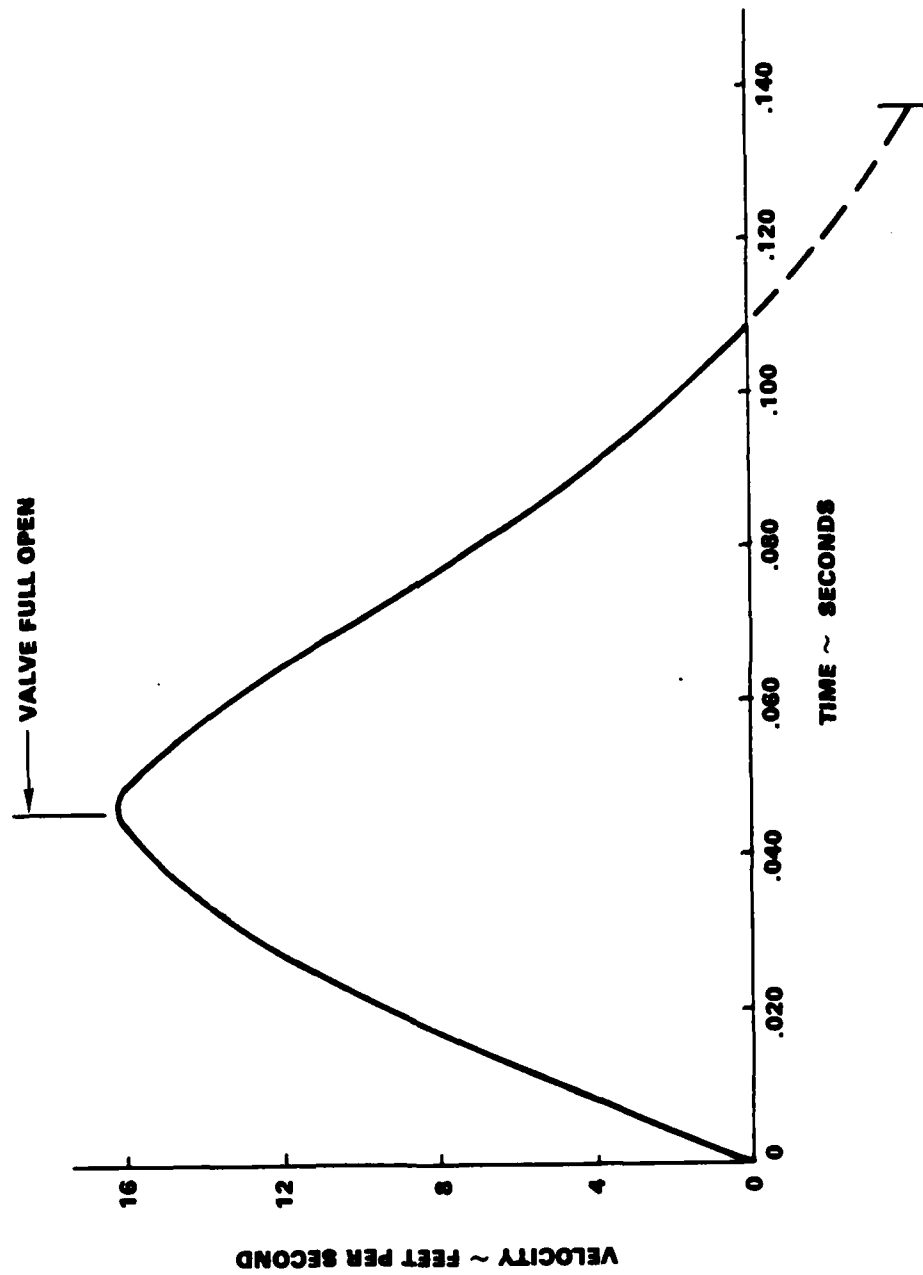Figure 11. Sleeve acceleration.

30

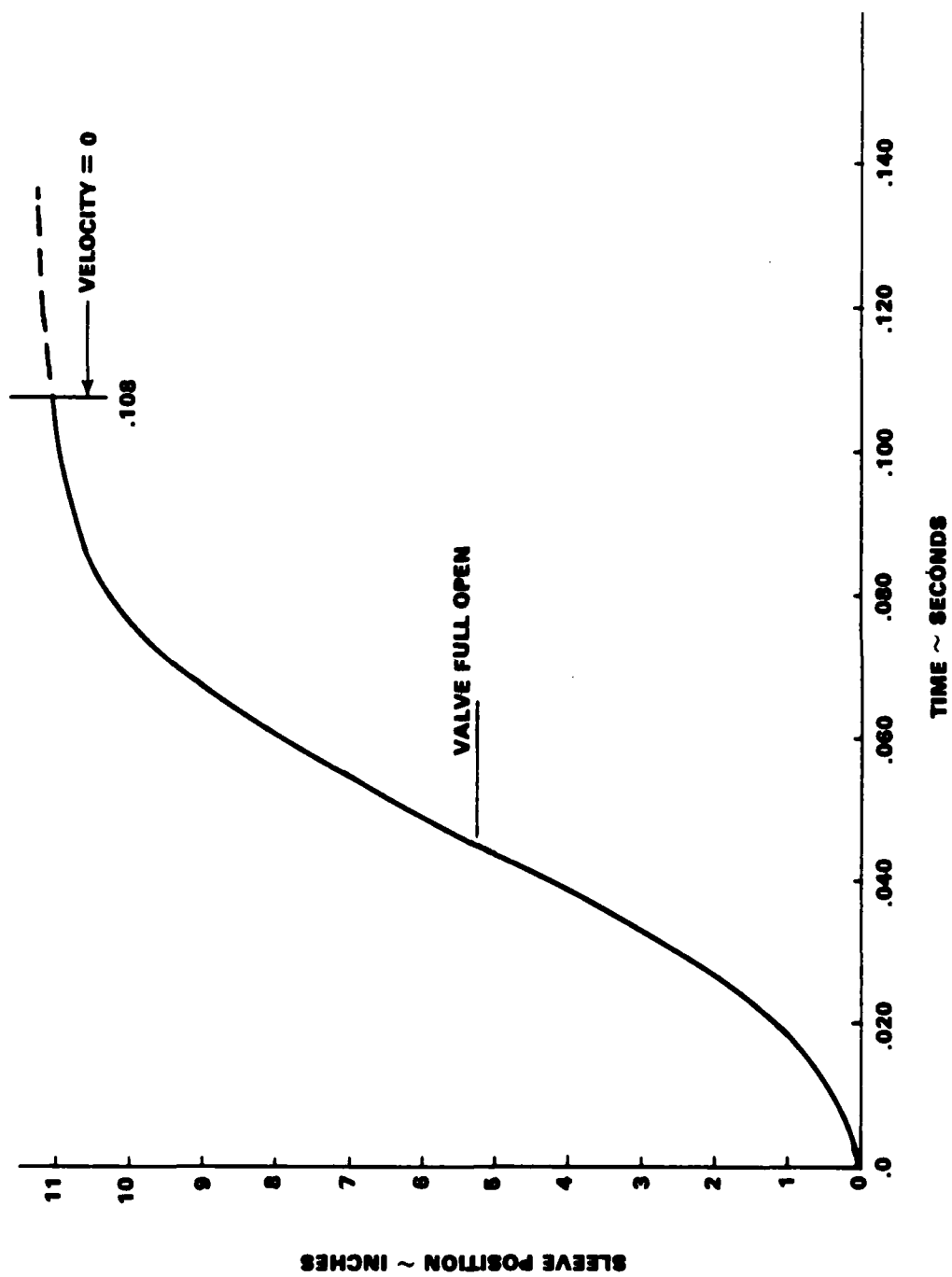Figure 12. Sleeve velocity.

31

Figure 13. Sleeve position.

assumed to be constant over the entire stroke. Just as there is a force reversal at 50 milliseconds there also is an acceleration reversal at this time. Sleeve velocity in Figure 12 is directly calculated from the acceleration data and shows a peak of 16 feet per second when the valve is fully open. Figure 13, showing sleeve position with respect to time, is based on the velocity curve and indicates that for the position at which the sleeve reaches full open (5.25 inches) the time is approximately 46 milliseconds. For the initial pressures assumed, the valve will open within the required 50 milliseconds. Figure 14 shows the amount of port area exposed as a function of time.

2.6 CONCLUSIONS

The conceptual design considered a single sliding sleeve valve size (36-inch diameter at 2500 psi) and focused on two critical areas of valve performance--the drive system and the sealing system.

In investigating the drive system, a complete hydraulic actuator circuit was designed and calculations of the expected performance characteristics of the valve were completed. Results of the calculations showed that the valve could be opened in less than 50 milliseconds and deceleration could be accomplished with comparatively moderate loads on the sleeve using conventional hardware.

The sealing system design revealed a potential problem with differential contraction of the shell versus sleeve; however, a satisfactory seal can be accomplished by reducing the clearances provided in the zero-pressure case. It will require additional calculations and, perhaps, some empirical studies to determine the optimum balance between clearances required for motion of the sleeve and those required for an effective seal. Another possible solution to the problem is to reduce the safety factor for shell design to better match the deflection to that of the sleeve.

Seal design at temperatures of 400°F or below can be easily accomplished using readily available elastomeric compounds. At temperatures of 650°F there are no commonly known compounds which are rated for satisfactory sustained service in a dynamic application. Some com-

33
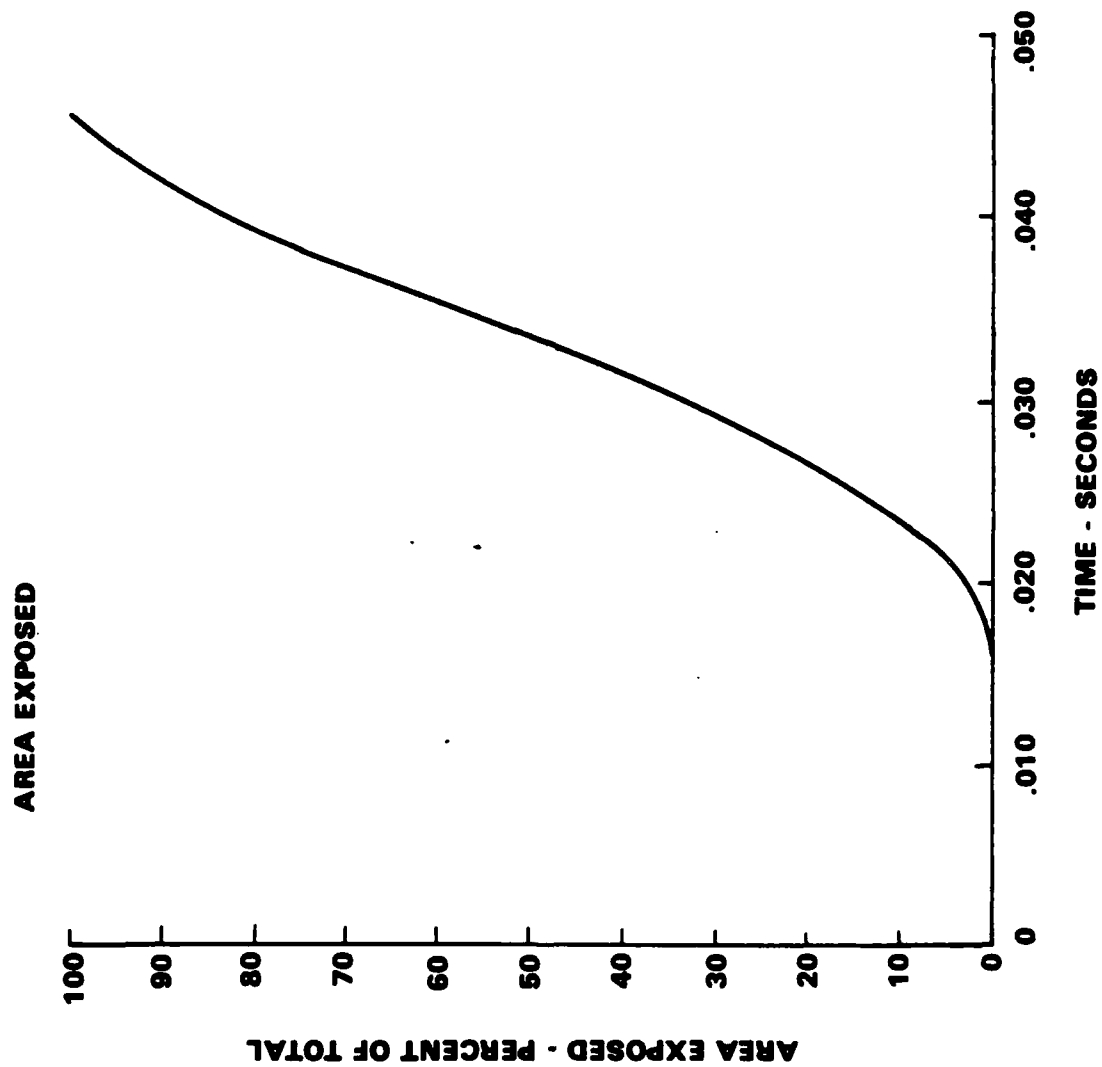
AREA EXPOSED

Figure 14. Port area exposed as a function of time.

34

pounds are rated for static seals approaching this temperature and may, on the basis of tests, prove to be adequate for low cycle life times.

The conceptual design revealed no significant problems with production of a valve of the size and pressure rating desired if operating temperatures do not exceed 400°F.

SECTION 3
PROTOTYPE VALVE TEST PROGRAM AND COSTS

Several sliding sleeve valves (possibly 20 or more) will be required for construction of the LB/TS facility. For this reason, preliminary testing should be conducted to verify expected valve performance characteristics before fabrication of the production valves. At the conclusion of final design, a valve development program should be initiated in which a full-scale prototype will be constructed and tested. A complete test program for the valve should explore valve opening time and how various factors such as charge pressure level, hydraulic system pressure levels, and seal friction affect the opening time. Seal performance as related to compound selection, surface finish of the shell, temperature effects, and other parameters should be evaluated.

A comprehensive test program requires several hardware items in addition to the prototype valve (see Drawing #M-1 for layout of test set-up). The prototype valve itself would include the flanged shell, sleeve, and hydraulic system including only the hydraulic cylinder, valving, and accumulators and pressure reservoirs. Additional hardware required includes a length of charge tube designed for 2500 psi. The length is determined by calculating the time required for the pressure wave generated when the valve opens to travel the length of the tube, rebound, and return to the valve opening. This time must be sufficient to allow the valve to open fully before the pressure wave reaches the valve opening in order to accurately model charge pressure effects on valve opening time (see Section 2.2 on "self-closing effect"). Also required is equipment to charge the hydraulic system consisting of nitrogen storage bottles, a nitrogen booster compressor, and a hydraulic power unit to fill the system and provide low-speed cycling of the sleeve. An air compressor is also required to operate the nitrogen booster. Pressurizing the large volume of the charge tube (approximately 500 cu. ft.) in a reasonable length of time requires a high-capacity compressor. It is proposed that a reciprocating air compressor or cryogenic compressor pumping a liquid gas be rented for this purpose. Instrumentation for the valve will be relatively simple. In order to measure the speed of valve opening, a linear potentiometer will be

attached to the hydraulic cylinder and a curve of sleeve position versus time will be obtained. This curve can be integrated to obtain sleeve velocity and integrated once again to obtain acceleration curves. A pressure transducer will be used to monitor pressure fluctuations near the valve opening to correlate their effect with valve opening time. Additional transducers will be mounted in the charge tube to allow the calculation of flow rate from the valve in order to determine the ideal ratio of port area to effective cross-sectional area of the valve. This data will be taken at various charge pressures to determine the effect of pressure level on this ratio. A data acquisition system capable of recording the output from the transducer and linear potentiometer at approximately one-millisecond intervals will be required as well as a switching system which will operate the solenoids of the hydraulic system valves and control the data acquisition system. Miscellaneous pressure gages, piping, and manual valving complete the hardware requirements.

The objectives of the study as previously stated include a study of valve opening time. Calculated valve opening time is within the 50-millisecond maximum requirement. These calculations are based on assumed seal friction values, hydraulic system pressure drops, and differential pressure effects on the sleeve. A series of tests will be conducted in which actual values for these quantities will be determined. The hydraulic system designed for the prototype is capable of producing a wide range of loading on the sliding sleeve and can therefore produce opening times less than the maximum for a variety of friction, pressure drop, and differential pressure effects. Adjustments will be made to the hydraulic system and dampers with the goal of reducing acceleration and deceleration loads on the sleeve to a minimum while maintaining the required 50-millisecond opening time. Charge pressure values versus opening time will be studied to determine what effect (if any) the charge pressure has on sleeve movement.

A critical area of valve operation is seal performance. The present design differs from the design of the successful pilot HIRT valves not only in size and pressure rating but also in the way in which pressure is carried by the shell (see discussion of seal design in

37

Section 2.4). This difference in shell loading causes differential contraction between the shell and the sleeve. The resulting space must be sealed by the O-ring. If sleeve fits are specified as the standard for a sliding fit, the clearance that results under pressure will be difficult to seal with the O-ring arrangement used in the pilot HIRT valves. The prototype valve will be constructed with the smallest clearance possible which will still allow a sliding fit. The valve will then be tested and, if friction loads are too high to achieve the desired opening time, the sleeve will be remachined to produce an optimum balance between friction loads and the ability of the O-ring to produce a static seal. Various lubricants will be tested to determine which produce the lowest friction loads.

Durability of the seal will also be studied during prototype testing. Experience with the pilot HIRT valves have shown that seal designs of this type can withstand the loadings developed in valve opening at pressures up to 1000 psi. The prototype testing will confirm the seal's ability to perform at 2500 psi. The effects of temperature on the seal will also be determined at the prototype stage. A nitrile compound (BUNA-N®) will be used for testing in the 200°-250°F range and silicone compounds will be used at higher temperatures. A goal of prototype testing will be to determine at what operating temperature available seal materials can provide acceptable durability.

It is expected during testing that component failures will occur in trying to achieve maximum valve performance. For this reason the costs generated for the test program include the following additional costs beyond the base equipment.

- Fabrication of one additional sleeve
- Purchase of an additional hydraulic drive system (including cruciform supports and installation)
- Two complete remachinings of the sleeve
- One complete remachining of the bore
- Removal of valve from test site and transportation to shop (four times)
- Installation of sleeve, seals, and cost of seal materials (four times)

In addition to these costs a complete test program at maximum operating temperature and pressure would require some special items which are not included in the estimate prepared herein. When testing at 2500 psi with a 30-ft. length of charge tube attached, the thrust loads developed at valve opening (2 million pounds) require a special foundation system. Very high sound levels are also generated which would require a silencer system if testing were done in a populated area. Finally, if precise high-temperature testing were performed, a temperature control and measurement system would be required. These three items--a special foundation, silencer system, and temperature control system--are not included in the estimate. It is assumed that all the hardware provided be designed for 2500-psi testing but that the fabricator of the valve only provide operational tests to 500 psi at essentially ambient temperature conditions. Structural "proof" tests only would be performed at 2500 psi. This would identify any basic problems in the design such that they could be rectified by the fabricator. The valve with all test hardware would then be shipped to a site of DNA's choosing where, presumably, foundations exist that would be suitable for full-pressure testing and are isolated to the point that a silencer system would not be necessary. The following tables give a cost breakdown as noted below:

Table 2 One full-size prototype valve - includes engineering costs and costs of a hydraulic system which includes the cylinder, accumulators, reservoirs, and associated valving. The complete valve is provided up to the driver tube transition piece which is not included.

Table 3 Test set-up costs - includes engineering and laboratory labor costs and costs of all hardware to perform tests to 500 psi as previously described. Includes transition piece, charge tube, and other items shown on Drawing #M-1 including one full-size prototype valve.

Table 4 Production valve costs - includes costs for 20 production valves with hydraulic system, including cylinder accumulators, reservoirs, and associated valving. The complete valve is included up to the transition piece which is not included.

Table 2. Prototype valve costs (FY85 dollars).

Activity			Cost
Engineering [includes all engineering efforts up to and including shop drawing preparation, procurement, and delivery of valve]			$ 53,250
Vavle Fabrication			
Labor			$ 298,000
Materials			122,000
	Subtotal		$ 420,000
	Total w/o Contingency	473,250	
	20% Contingency		94,650
	Total		$ 567,900

Table 3. Test set-up costs.
(includes one prototype valve)

Activity			Cost
Engineering [includes all engineering efforts for design and procurement of a prototype valve and all laboratory costs for a test and development program resulting in a final valve design]	Prototype Valve Development Costs	67,750	$ 53,250
	Subtotal		$ 120,900
Valve Fabrication			
Labor			$ 298,000
Materials			122,000
	Subtotal		$ 420,000
Test Equipment			
Labor			$ 200,600
Materials			149,400
	Subtotal		$ 350,000
	Total Cost w/o Contingency	$ 890,900	
	20% Contingency		178,200
	Total		$1,069,100

Table 4. Production valve costs.
(for 20 production valves)

Activity	Cost
Engineering	$ 18,900
[includes all engineering costs up to and including closing out the fabrication contract]	
Valve Fabrication	
Labor	$5,364,000
Materials	2,196,000
Subtotal	$7,560,000
Total w/o Contingency	7,578,900
10% Contingency	757,900
Total	$8,336,800

42

APPENDIX

DRAWINGS

43

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